Boiling-Film Heat Transfer Coefficients in a Long-Tube Vertical Evaporator

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Boiling-Film Heat Transfer Coefficients in a Long-Tube Vertical Evaporator

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THE long-tube vertical evaporator is of the natural circulation type but is distinguished from the ordinary apparatus of this type by the length of the tubes used, which are sometimes as much as 22 feet. The liquid is fed into the bottom of the tubes and is propelled upward at a rather high velocity by the vapor evolved from the boiling liquid. In this way the advantage of high velocities, found in forced-circulation evaporators, is combined with low power requirements of natural circulation evaporators, and these advantages make this type economical and practicable for the evaporation of many kinds of liquids.

Although many papers have appeared in the literature on this type of evaporator, they are mostly of a qualitative nature. A recent paper by Brooks and Badger (*2*), however, presented the results of a thorough investigation on heat transfer coefficients in a long-tube vertical evaporator. They divided the tube into a boiling and a nonboiling section, studied each section separately, and presented a correlation of the over-all heat transfer coefficients in the boiling section of the tube. Previous investigators on boiling in evaporator tubes (*3, 18, 19, 20*) had treated the tube as a whole, and the coefficients presented were not true boiling coefficients but included the effect of a portion of the tube in which the liquid was not boiling.

In the present investigation, the nonboiling section was eliminated entirely by having the liquid under boiling conditions throughout the entire length of the tube. Tube wall temperatures were measured so that the individual film coefficients as well as the over-all heat transfer coefficients could be determined. The results presented in this paper deal with the boiling film coefficients only.

Until recently very little work had been done on the study of the mechanism of boiling, though in the past few years rapid developments on the subject have been made. Some of the most thorough investigations of boiling have been carried out by Jakob and his co-workers, notably Fritz and Linke (*7, 8, 11–15*). These workers studied the problem from a theoretical as well as experimental standpoint, using both horizontal and vertical heating surfaces; many of their results were based on actual photographic measurements.

A number of recent experimenters have determined film heat transfer coefficients to boiling liquids (*3, 4, 5, 14, 15, 16, 18, 19, 20*), but as yet no satisfactory general equation has been developed for predicting these coefficients. There are wide variations in the numerical values found, and the results are usually applicable only to the type of apparatus used. The major variables affecting the coefficient are the temperature difference from the surface to the liquid, the temperature of the liquid, the nature and condition of the surface, and the wettability of the surface by the liquid, though there is considerable variation in the influence attributed to each of these variables by the different investigators.

Data on film coefficients to boiling liquids in vertical tubes are comparatively limited (*1A, 3, 18, 20, 22*). The coefficients

Film coefficients are given for water, sugar, and "Duponol" solutions, boiling in a long-tube vertical evaporator, equipped with a single 20-foot tube, under a wide range of conditions. From the data obtained, an empirical correlation of the coefficients is derived, expressing the coefficients in terms of the Prandtl number, surface tension of the liquid, specific volume of the vapor, and average temperature drop across the liquid film; all of these are based on the average temperature of the boiling liquid.

A relation between the average liquid temperature and the vapor temperature is derived, from which the average liquid temperature can be predicted from variables usually known to the commercial designer.

An explanation for the effect of the several variables is given in the light of this and previous investigations, and of certain known factors affecting the mechanism of boiling in the tube.

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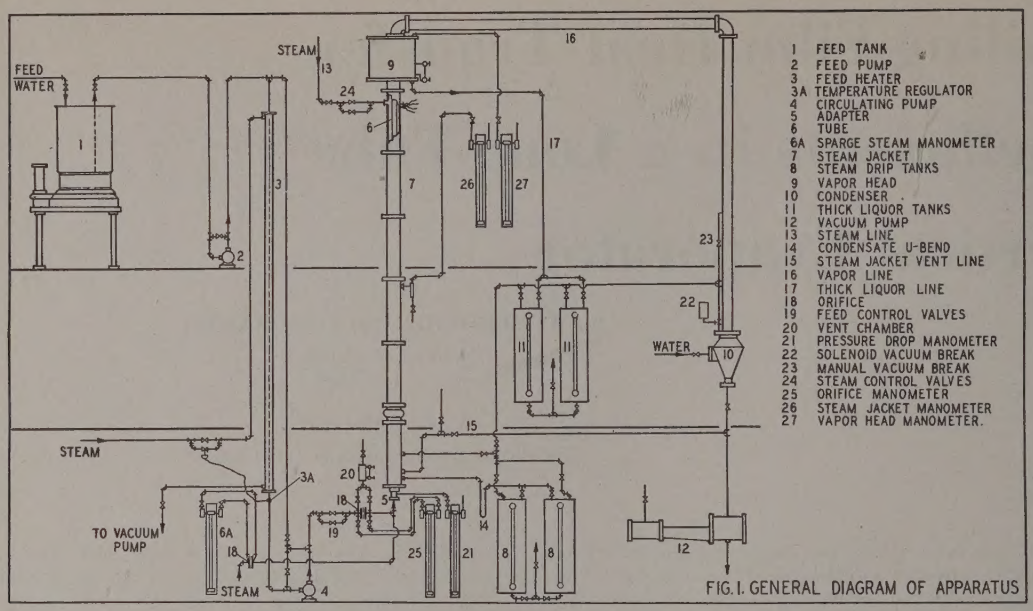

FIG. I. GENERAL DIAGRAM OF APPARATUS

have been found to be a function of the temperature difference, the temperature of the liquid, the level of the liquid in the tube, and its rate of circulation. It is generally agreed that boiling coefficients are considerably higher than nonboiling coefficients.

Apparatus

The apparatus used in this investigation consisted of a single-tube evaporator of the long-tube vertical type, with the usual auxiliary equipment, and various necessary attachments for controlling and measuring the conditions of operation. The tube was made of copper, 20 feet in length and 2.00 inches o. d. × 1.76 inches i. d. Figure 1 shows a diagram of the apparatus:

Feed water or solution was drawn from the feed weigh tanks, 1, by pump 2, and pumped through heater 3 and into the bottom of tube 6. Another pump, 4, recirculated part of the feed through the heater; a temperature regulator, 3A, on the exit line from the heater, controlled a valve on the steam line to the heater. The feed rate was controlled by valves 19 and was measured by means of an orifice, 18, connected to a mercury manometer, 25. Open steam could be injected into the feed just as it entered adapter 5, and was measured by orifice 18 and manometer 6A. Steam was fed to the insulated steam jacket, 7, through valves 24, and its pressure was measured by manometer 26. The boiling liquid, heated by the steam in the jacket, passed up the tube and was ejected at the top into vapor head 9, where the liquid and vapor were separated. The vapor passed through line 16 and into condenser 10, from which the mixture of condensed vapor, condenser water, and noncondensed gases was withdrawn by wet vacuum pump 12. Pressure in the vapor head was measured by manometer 27 and was controlled by means of an automatic solenoid valve, 22, as well as the manually operated valve, 23. The remaining liquid in the vapor head was withdrawn through line 17 into measuring tanks fitted with gage glasses. The pressure in these tanks was equalized with the vapor head to permit the flow of liquid through line 17. Steam condensate was removed through the U-bend, 14, and collected in drip tanks 8, also provided with gage glasses.

All temperatures were measured by means of copper-constantan thermocouples. These were placed in suitable positions to measure the temperatures of the feed, thick liquor, steam in the steam chest, vapor head, steam condensate, sparge steam, tube wall, and liquid in the tube. The tube wall temperatures were measured by thermocouples, embedded in the tube wall by the method of Hebbard and Badger (*10*) at intervals of one foot. The temperature of the liquid in the tube was measured by means of a traveling thermocouple similar to that described by Brooks and Badger (*2*), which could be moved longitudinally along the tube to measure the temperature at any point. These thermocouples were calibrated in place against reference thermocouples which had previously been calibrated in an oil bath.

All other thermocouples were made of heavily insulated No. 24 Birmingham wire gage, copper-constantan wire, with the leads sealed into short lengths of $^3/_8$-inch o. d. copper tubing so that the junction projected a short distance beyond the end of the tube. These were calibrated in an oil bath against a platinum resistance thermometer, and could be inserted in their respective positions through packing glands for easy removal. The electromotive forces of the thermocouples were measured by a Leeds & Northrup portable precision potentiometer.

Pressures in the steam chest and vapor head were measured by ordinary laboratory mercury manometers. The pressure in the steam chest was controlled by a valve on the steam line, and the vapor head pressure was controlled by a valve on a vacuum break entering the vapor line just above the condenser. Both of these valves were manually operated. For further control of the vacuum, a solenoid vacuum break was used, which operated automatically in connection with the vacuum manometer and controlled the pressure to within ±1 mm. of mercury.

In order to ensure boiling at the bottom of the tube, the feed was heated 5° to 10° F. above the boiling point in the vapor head, and in addition a certain amount of sparge steam was injected into the bottom of the tube. The amount of this sparge steam was measured by an orifice in the steam line. Before starting a run, a few readings of the traveling thermocouple were made near the bottom of the tube to determine if boiling actually existed there, as indicated by the maximum liquid temperature being at the bottom.

Experimental Procedure

A run consisted of maintaining constant conditions, after a steady state had been attained, for a period of 20 minutes, during which time the necessary data were recorded. Three operators were required for successful operation of the machine and the recording of data, which included readings of the steam drip and thick liquor tanks, initially, finally, and at 4-minute intervals; readings of all manometers at 2-minute intervals; three readings of each thermocouple; and three complete traverses of the traveling thermocouple along the length of the tube.

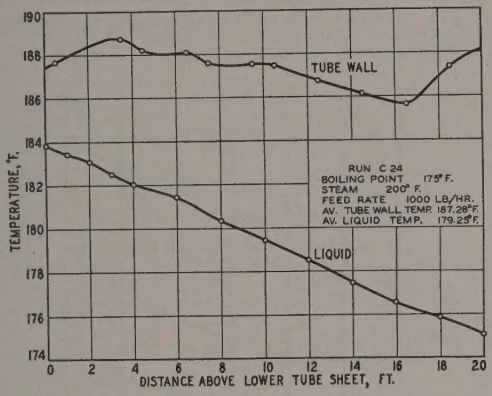

FIGURE 2. TYPICAL DISTRIBUTION OF TUBE WALL AND LIQUID TEMPERATURES

Since the feed temperature necessarily depended on the boiling point of the liquid, there were only three independent variables. These were (*a*) the boiling point in the vapor head, which was varied from 150° to 200° F., (*b*) the over-all temperature difference, varied from 10° to 60° F., and (*c*) the feed rate, varied from 250 to 2,000 pounds per hour.

Constancy of surface conditions is apparently important in obtaining uniform results, and therefore to ensure a clean surface at all times, the inside of the tube was boiled out every 3 or 4 days with a dilute solution of inhibited hydrochloric acid.

The thermocouple readings used were the averages of the three observed readings for each couple. The average liquid temperature was determined by plotting the averaged readings of the traveling thermocouple in millivolts against the distance from the bottom tube sheet. The resulting curve was then integrated with a planimeter, and the area under the curve divided by the length of the abscissa. This gave an integrated average millivolt reading which was then converted to degrees Fahrenheit. A similar procedure was followed in calculating the average tube wall temperatures. In averaging millivolts instead of actual temperatures, any error introduced would be negligible, since the calibration curve converting millivolts to temperatures is essentially a straight line over the range of temperature variations along the tube.

Typical distribution curves for the tube wall and liquid temperatures are shown in Figure 2. The liquid film temperature drop, used for calculating the film coefficients, was the difference between the average tube wall and the average liquid temperatures. This temperature difference was corrected for the small temperature drop across that portion of the tube wall between the thermocouple junction and the inside surface of the tube.

The quantity of heat transferred through the tube wall was determined from the amount of steam condensate collected, and the known temperature and pressure of the steam in the steam chest, radiation being neglected. Entrainment of liquid with the vapor from the vapor head prevented a close check of the heat input and output, but the differences between the two remained fairly consistent.[2]

Correlation of Boiling Film Heat Transfer Coefficients

It was believed that the most important variable affecting the film coefficients would be the liquid film temperature drop, Δt_L. Accordingly, when sufficient data had been taken, plots were made of the film coefficients against liquid film temperature drops for each boiling point and each feed rate. Runs were made with water at 250, 500, 750, 1,000, and 2,000 pounds per hour, and at each feed rate three boiling points were used, 150°, 175°, and 200° F. The plots for the 500-pound-per-hour feed rate are shown in Figure 3; these curves are representative of the other feed rates as well. Each curve apparently passed through a minimum, and the coefficients were generally lower at the lower than at the higher boiling points; other experimenters (*4*, *16*, *19*) have also observed this.

It was soon found that by means of a simple viscosity function, a fairly good correlation was obtained which fitted almost all of the runs with water within 20 per cent. However, it was obvious that other factors besides viscosity must be involved in any change in the film coefficient, although with the use of water alone the changes in other variables were so small as to be almost negligible.

In order to determine more accurately the extent to which viscosity entered the correlation, it was considered necessary to give this variable a wider range in the experimental work than could be obtained with the changes in temperature of water alone. Accordingly, a few supplementary runs were made; varying concentrations of sugar solution were used up to 50 per cent. This gave approximately a tenfold increase in the viscosity, over the minimum obtained with water. Also, in order to determine the effect of surface tension, several more runs were made in which a small amount of "Duponol," a soapless detergent, was added to the feed water, lowering the surface tension almost 50 per cent.

The results of these experiments were very surprising. The boiling film coefficients obtained with the "Duponol" solutions were two to four times higher than with pure water,

[2] The data obtained are summarized in a very long table. Since the table will be given in full in the February 25, 1939, issue of *Trans. Am. Inst. Chem. Engrs.*, it is not included here.

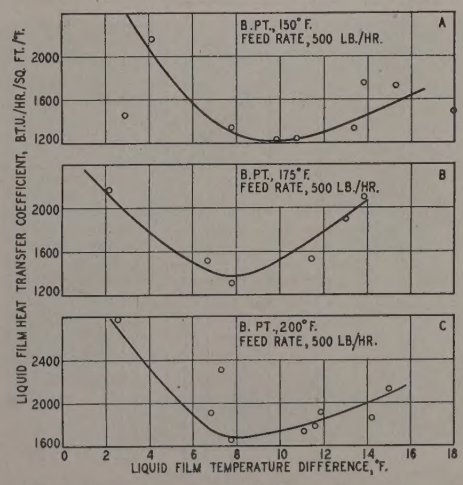

FIGURE 3. PLOTS FOR FEED RATE OF 500 POUNDS PER HOUR AT VARIOUS BOILING POINTS

under the same conditions. For example, with a feed rate of 750 pounds per hour, a boiling point of 200° F., and a 35° F. over-all temperature drop, the boiling film coefficient for water would be about 1,800 B. t. u. per hour per square foot per ° F.; with "Duponol" solution, the coefficient under the same conditions was over 5,400.

These experiments indicated that the surface tension had a definite influence on the mechanism of boiling in the tube and should be included in any correlation of boiling film coefficients. The thermal properties of the liquid were introduced in the form of a Prandtl number, and a function of vapor specific volume was also used, since this factor varied almost threefold over the temperature range used and would therefore affect the velocity in the tube. The proper functions of these several variables were found empirically, and the entire function,

$$\frac{h_L \left(\frac{C_p \mu}{k}\right)^{0.3} \sigma^2}{v^{0.1} (\Delta t_L)}$$

when plotted against Δt_L produced a band of points which could be represented by a straight line (Figure 4). From this can be derived the following equation:

$$h_L = \frac{7.8 \times 10^6 \times v^{0.1}}{\left(\frac{C_p \mu}{k}\right)^{0.3} \sigma^2 (\Delta t_L)^{0.13}} \qquad (1)$$

Approximately 70 per cent of the runs made fall within ±15 per cent of the values calculated from this equation, and 90 per cent within ±20 per cent of the calculated values.

Equation 1 does not include any factor which would take into account a change in the length and diameter of the tube, and the data do not justify the inclusion of such a term in this empirical equation, since these factors were kept constant during all of the experimental work. However, recalculation, on the basis of Equation 1, of some of the data of Brooks and Badger (*2*), with boiling lengths varying from 12 to 18 feet indicate a slight trend towards lowering of the coefficient with a decrease in the boiling length, for tubes of the same diameter.

Equation 1 is entirely empirical and therefore should be used with discretion for conditions appreciably divergent from those covered in this work. However, since the factors involved were varied over a fairly wide range, it is believed that the equation can be safely used for various solutions under conditions similar to those in this investigation.

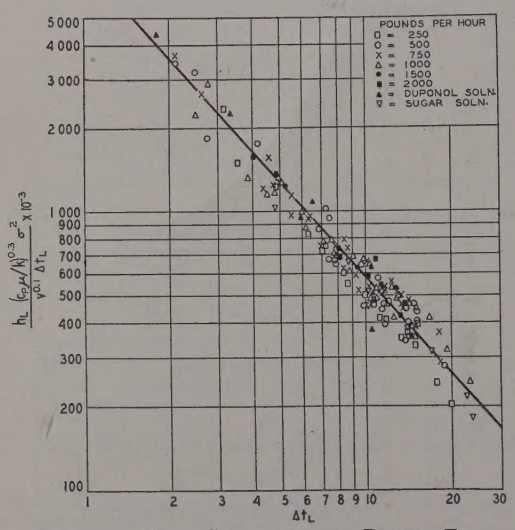

FIGURE 4. FINAL CORRELATION OF BOILING FILM COEFFICIENTS

Correlation of Liquid and Vapor Temperatures

The variables in Equation 1 are all based on the average liquid temperature, and consequently the equation and Figure 4 are applicable only when this temperature is known. Therefore, in order to make Equation 1 practicable for design work, it is necessary to have some relation between the average liquid temperature and the independent variables, or those known to the commercial designer. These factors are (*a*) the over-all temperature drop from steam to vapor head, ΔT_0, (*b*) the vapor head temperature, t_v, and (*c*) the feed rate, w.

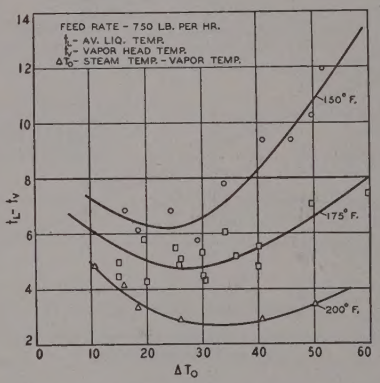

FIGURE 5. DEVIATION OF AVERAGE LIQUID AND VAPOR TEMPERATURE

The temperature of the liquid in the tube is always higher than that in the vapor head, as indicated in Figure 2. This is due to the fact that the liquid is boiling throughout the entire length of the tube and is therefore at the saturation temperature; this saturation temperature is always higher than the vapor head temperature, because there is a pressure decrease in progressing upward along the tube. The pressure drop across the tube is the sum of several individual pressure drops as follows:

The static pressure drop, determined by the weight of the liquid-vapor mixture in the tube.
The pressure loss due to the acceleration of the liquid and vapor to the outlet velocity.
Other pressure drops, such as friction loss and outlet loss.

The total pressure drop, and thus the liquid temperatures, will depend on the extent of each of these pressures. At low values of ΔT_0 there will be a large proportion of liquid in the tube, and the static head will predominate. As ΔT_0 is increased, however, more of the liquid will be evaporated, and the remaining liquid will be ejected from the tube at a higher rate, as a result of the large increase in volume. This will cause a decrease in the static head and an increase in the acceleration and friction heads. As ΔT_0 is increased further, this trend will continue until the latter pressures are predominating. Thus, when ΔT_0 is increased, it would be expected that the liquid temperatures will decrease at first to a minimum and then rise again, forming U-shaped curves. This effect should be the same at all vapor head temperatures, except that the average liquid temperatures at a given ΔT_0 should differ as a result of variations in such factors as the heat transfer coefficients and specific volume of the vapor with t_v. Figure 5 shows the variations of $(t_L - t_v)$ with ΔT_0 at the three vapor head temperatures used and at one feed rate

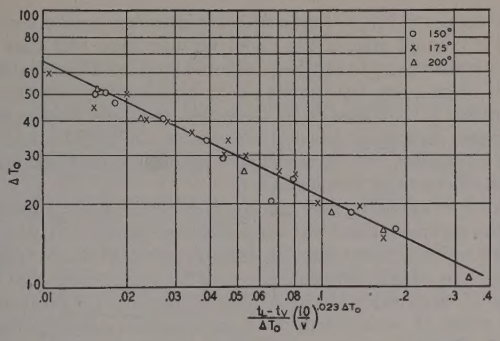

FIGURE 6. CORRELATION OF AVERAGE LIQUID TEMPERATURES AT FEED RATE OF 750 POUNDS PER HOUR

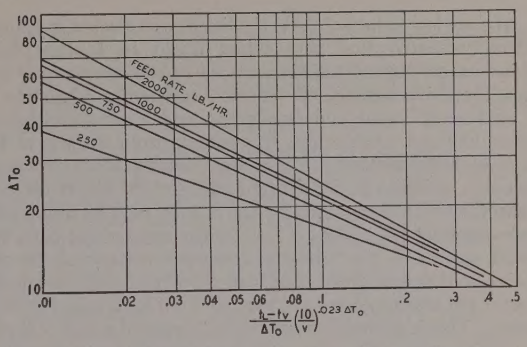

FIGURE 7. CORRELATION OF AVERAGE LIQUID TEMPERATURES AT VARIOUS FEED RATES

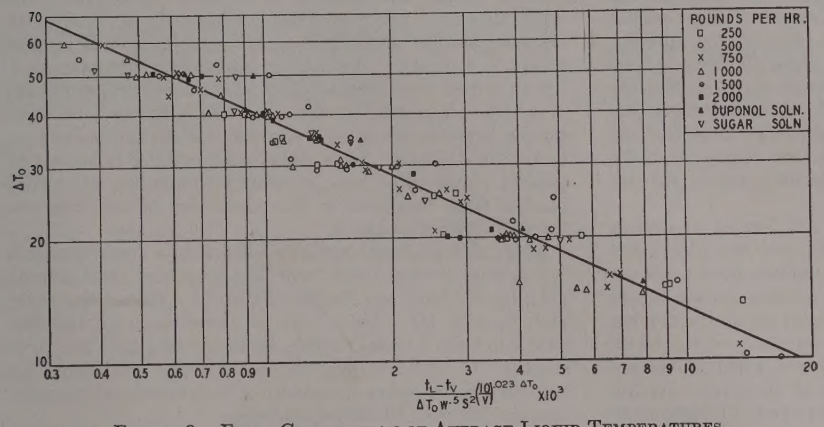

FIGURE 8. FINAL CORRELATION OF AVERAGE LIQUID TEMPERATURES

(i. e., 750 pounds per hour). Other feed rates gave similar sets of curves.

It was decided that the specific volume of the vapor was the major factor affecting the position of the curves at the different temperatures, as this property varied over almost a threefold range between the minimum and maximum temperatures used. However, the actual amount of vapor produced would also exert some influence. Although this quantity of vapor is usually not known, it is a direct function of ΔT_0 which can be used in its place. Therefore, to correct for the temperature in the vapor head, the specific volume, v, of the vapor at that temperature would have to be taken into account, and this would have to be expressed as some function of ΔT_0. The correct function was found to be $(v/10)^{0.023\Delta T_0}$. Figure 6 shows $\frac{t_L - t_v}{\Delta T_0}\left(\frac{10}{v}\right)^{0.023\Delta T_0}$ plotted against ΔT_0, for a feed rate of 750 pounds per hour. The points justify representation by a straight line. Similar curves were found for the other feed rates summarized in Figure 7.

A correction for variation in feed rate was accomplished by dividing the entire function by $w^{0.5} S^2$. Figure 8 shows a final correlation of all the runs. This consists of a band of points which can be represented by a straight line. When solved for t_L the equation for this line gives

$$t_L = t_v + \frac{2.8 w^{0.5} S^2 (0.1\, v)^{0.023\Delta T_0}}{(\Delta T_0)^{1.15}} \qquad (2)$$

From a practical standpoint it would probably be just as convenient to use the curves of Figures 5 or 7 to determine the average liquid temperature. The final correlation of Figure 8, however, has the advantage that it can be expressed in the form of a single equation.

Discussion of Results

Recent papers (*2*, *18*) described the mechanism of boiling in a vertical tube. Three different types of boiling have been observed, a homogeneous mixture of liquid and vapor in the form of a foam, alternate slugs of liquid and vapor, and "liquid film" boiling, in which the inner surface of the tube is covered with a film of liquid and with a core of vapor and fine spray moving up the center of the tube.

All three types of boiling probably occurred in the evaporator used in these experiments, but because of the conditions under which it was run, it is believed that the third type (i. e., film boiling) occurred over a major portion of the tube. This is substantiated by observations of the liquid and vapor mixture being ejected at the top of the tube which could be seen through sight glasses suitably placed in the vapor head.

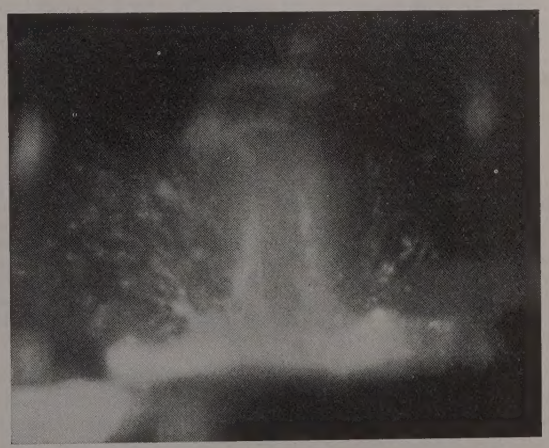

FIGURE 9. LIQUID LEAVING THE TOP OF THE EVAPORATOR TUBE

At very low values of ΔT_0, the liquid was ejected in spurts, which indicated that slug action might be taking place. However, during most of the runs the action was more or less steady and continuous. A column, apparently consisting of a mixture of vapor and fine spray, and approximately the same diameter as the tube, was coming from the top of the tube and the deflector. This was moving at a very high velocity, as indicated by a bright circle on the otherwise dark deflector which was caused by the rapidly moving liquid particles striking the deflector. A coarser spray came from the inside surface of the tube and moved at a much slower velocity, so that sometimes it did not even rise as high as the deflector but was projected outward and fell back onto the tube sheet. This is probably what would happen if a film of liquid were rising along the tube wall at a velocity lower than that of the central core, and indicates the presence of such a film on the inside surface of the tube. At high values of ΔT_0, the amount of coarse spray diminished and in some cases entirely disappeared, suggesting that the film became thinner towards the top and, under some conditions, disappeared entirely, leaving part of the tube surface dry. Figure 9 shows typical action at the top of the tube. The small traveling thermocouple tube extends through the gland on the bottom of the deflector and passes into the top of the larger tube, and the column of vapor and fine spray can be seen coming from the top of the tube and striking the deflector, causing the appearance of a bright spot.

According to Jakob (*11*), bubbles are formed at certain definite spots on the heating surface, and the higher the temperature drop, the more of these nucleus points will occur. A bubble thus formed on this heating surface would grow until it was large enough to be swept off by the moving liquid film or until it burst through the film. In the latter case some small droplets would be formed, which would account for the presence of the fine spray observed. As the Δt_L is increased and more bubbles are evolved, it is reasonable to assume that a larger portion of the surface will be covered with vapor and less will be wetted by the liquid, and this is probably the reason for the slight decrease in the coefficient with an increase in Δt_L, as represented by the negative exponent of this factor in Equation 2. An increase in Δt_L would also increase the velocity in the tube, which would tend to counteract the above effect, and the combined effects are probably the reason for the small influence exerted by Δt_L.

Mueller (*21*) has correctly suggested that the apparent near independence of h_L of Δt_L may be due to the fact that the variation of Δt_L from one end of the tube to the other is large as compared to the range of the mean values of Δt_L for the entire tube, as these have been here calculated. He points out that a more correct procedure would be to evaluate the coefficient from the expression

$$\int dq = \pi D \int \Delta t_L h_L dL$$

Experimental data permit evaluation of Δt_L in terms of L but unfortunately do not permit a corresponding evaluation of h_L. The assumption that h_L varies as an exponential function of Δt_L seems unwarranted, since this would ignore the effect of the increased velocity of liquid due to the increased volume of vapor as the liquid progresses up the tube. Nevertheless, if such an assumption is made, a few calculations seem to confirm Equation 1 in the sense that h_L so determined varies as Δt_L to a small negative exponential power.

Few writers have mentioned the significance of surface tension on the heat transfer coefficient. Jakob and his coworkers, however, showed both experimentally and theoretically that surface tension is an important factor in heat transfer to boiling liquids, both as affecting the size of the bubbles and the wettability of the heating surface.

Fritz showed (*7*, *8*) that a decrease in surface tension reduces the maximum size of the vapor bubbles, and also that the heat transfer coefficient from a liquid to a bubble of vapor is very large when the bubble starts and decreases as its size increases. It follows from these two relations that, with a decrease in surface tension, the size of the bubbles evolved will be smaller, and consequently a greater average heat transfer coefficient can be expected.

An increase in the heat transfer coefficient due to a lowering of the surface tension has been observed before (*13*), but the effect was much less than has been observed in this investigation. Apparently the conditions of boiling occurring in this type of evaporator are ideal for the maximum utilization of the effect of surface tension in increasing the coefficients to boiling liquids.

Another important factor in boiling is the wettability of the surface by the liquid. This has a marked effect in that it governs the size and shape of the bubbles and the rate with which they are evolved from the heating surface. This factor has been studied and explained in detail by several investigators (*6*, *8*, *11*, *13*). Wettability is related to surface tension, but it cannot be expressed as a function of surface tension alone since it is also closely connected with the interfacial tension between the liquid and metal and the surface tension of the metal itself. Since practically nothing is known at present about these two, an empirical function of surface tension alone was used in the correlation of the data obtained in this investigation.

Methods for calculating film coefficients for steam condensing on long vertical tubes have been presented by Kirkbride (*17*), by Hebbard and Badger (*9*), and by Baker, Kazmark, and Stroebe (*1*). Using one of these methods together with Equation 1, it is possible, by means of a trial and error solution, to calculate over-all heat transfer coefficients for this evaporator from a knowledge of the vapor head temperature and the over-all temperature drop.

Conclusions

1. This investigation, to the authors' knowledge, is the first systematic work to have been done on liquid film coefficients in a long-tube vertical evaporator, in which the boiling occurs over the entire length of the tube. The heat transfer coefficients found across the boiling film were higher than expected and in all cases were considerably higher than the steam film coefficients.

2. An equation has been presented which can be used for predicting boiling film heat transfer coefficients in an evaporator of this type, and operated under conditions similar to those used in this investigation.

3. A relation has been found between the average liquid temperature and the vapor head temperature, from which it is possible to predict the average liquid temperature from variables that are known to the commercial designer. This relation is important for the practical application of the heat transfer coefficient correlation, since the latter is based on the average liquid temperature, a factor which is seldom known in design work.

4. The observed behavior of the coefficients and temperature distributions has been explained in the light of a proposed mechanism of boiling in the long vertical tube, based on factors characteristic of this type of boiling.

5. The surface tension of the liquid and the ability of the liquid to wet the heating surface apparently have a strong influence on the heat transfer coefficient to boiling liquids, at least in this type of evaporator; and the increase in the coefficient due to a lowering of the surface tension was found to be much greater than had previously been observed or generally recognized.

Nomenclature

Cp = specific heat at constant pressure
D = diameter, ft.
h_L = boiling liquid film heat transfer coefficient, B. t. u./(hr.)(sq. ft.) (° F.)
k = thermal conductivity, B. t. u./(hr.)(sq. ft.)(° F./ft.)
L = length of tube, ft.
q = heat transferred per unit time, B. t. u.
S = specific gravity
t_L = average liquid temperature, ° F.
t_v = vapor head temperature, ° F.
Δt_L = liquid film temperature difference, ° F.
ΔT_0 = over-all temperature difference = $t_s - t_v$
v = specific volume, cu. ft./lb.
w = weight of feed, lb./hr.
σ = surface tension, dynes/cm.
μ = viscosity, lb./(ft.)(hr.)

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